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HC condensate sphere inlet pipe failure investigation

During a normal turnaround and inspection (T&I) of a hydrocarbon sphere, a fatigue failure in the 24-in. inlet pipe riser was discovered. As per the original 2003 design, the sphere serves as a surge tank during pipeline scraping, as well as a backup for a stabilizer during T&I. Each stabilizer has a capacity of 78,000 bpd. However, due to asymmetry in the piping configuration, the sphere typically receives higher flow of 90,000 bpd, with rates sometimes reaching 110,000 bpd as a result of scraping or other activities happening concurrently. The sphere was upgraded to work continuously in 2012 to handle the anticipated additional condensate by adding an overhead gas compressor and a dedicated condensate shipper pump.

The surge sphere (**FIG. 1**) operates as a three-phase separator to separate water, gas and HC condensate. The HC condensate is stabilized at 50 psig in the surge sphere with the flashed vapor flowing to the overhead gas compressors. The stabilized condensate is pumped to the suction of the condensate shipper pumps, while sour water is pumped to sour water strippers.

The sphere has an inlet standpipe extended inside the sphere from the bottom to a height of 46 ft, with a U shape to direct the feed flow above the high-high level (HHL) of the sphere. The standpipe is mainly intended to provide the wet condensate adequate residence time inside the sphere for separation before it leaves from the bottom nozzle. This standpipe was found to be broken near the bottom nozzle (**FIG. 2**) after opening the sphere for a planned T&I. The support flange near the top of the sphere, shown in **FIG. 3,** was found detached with one broken bolt inside the sphere. The seven other bolts for the flange were missing, and it is likely these were carried into the suction strainers of the condensate and water pumps.

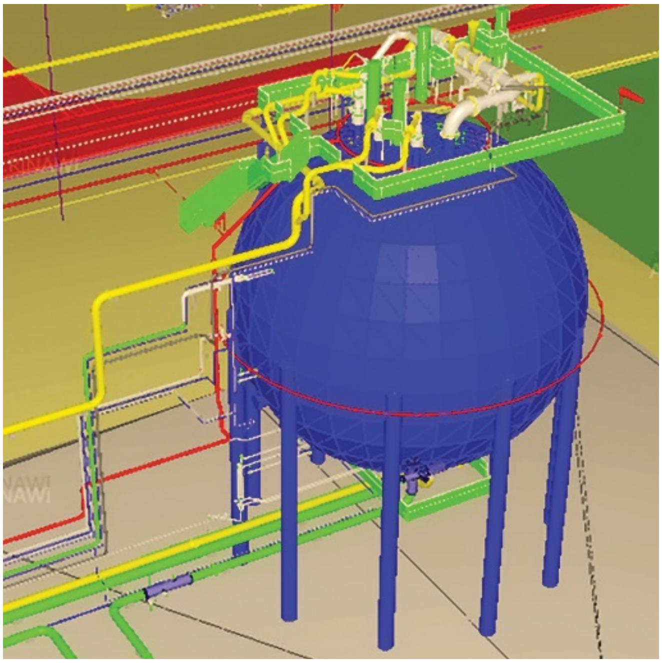

FIG. 1. The surge sphere operates as a three-phase separator to separate water, gas and HC condensate.

FIG. 2. Broken standpipe near the bottom nozzle.

History. The plant reported high vibration on the inlet HC condensate line below the sphere, as well as on the outlet gas line on top of the sphere. Two vibration surveys were conducted on the sphere and its piping. A detailed finite element stress evaluation of the sphere's external nozzles based on measured vibration showed that these nozzles can safely handle the vibration. After the inlet pipe was found detached inside the sphere (**FIG. 4**), further review of the process data was conducted. Findings included:

- A high condensate flowrate to the sphere beyond the original design rate of 2,275 gpm (78,000 bpd) was observed on several occasions and increased in frequency during October 2017–March 2018 (**FIG. 5**). The flow reached 3,500 gpm (120,000 bpd), particularly in October 2017. The average flow in that month was 3,200 gpm (110,000 bpd).
- Significant fluctuation in the sphere level was observed during the same period, which indicates sudden liquid surge to the sphere.
- The high flow to the sphere took place during the outages of one stabilizer, which was encountered more frequently in 2017–2018 due to stabilizer reboiler cleaning activities.
- The observed surge in flow to the sphere is caused by diverting excessive HC condensate to mitigate capacity limitation in the stabilizers, which may have also coincided with pipelines scraping operations.
- When one stabilizer is out for cleaning, the sphere is used as a backup stabilizer. During this scenario, the piping configuration preferentially diverts more flow to the sphere, causing the flow to exceed sphere design capacity.

The plant indicated that high vibration was observed during recycling of the HC condensate using the newly added 2,692-gpm pump with high discharge pressure. This scenario was investigated to check if the jet flow from this recycle 8-in. line would hit the failed inlet riser pipe.

Jet impact from condensate recycle line. It was reported that the sphere had visible high vibration during recycling condensate through the newly added pump with higher discharge pressure. The recycle 8-in. line enters the sphere near the top and is directed towards the center of the sphere. Jet flow was analyzed to see if the recycle line jet would hit the inlet pipe. **FIG. 6** shows the jet flow trajectory calculations. The calculated path shows that the jet has little effect on the inlet pipe.

Plug flow in two-phase, 24-in. condensate inlet. The 24-in. inlet pipe contains two-phase flow. Two-phase flow forces were calculated, assuming a condensate liquid flow of 3,200 gpm (110,000 bpd) and 16 MMft^{3}d of gas. These represent the flow in November 2017.

The maximum slug force is calculated as 2,376 lbf (pound-force), as shown in Eq. 1. It was assumed that the liquid slug is driven by the high-velocity gas into the sphere. The change in direction causes a large force due to momentum change. The force is multiplied by a factor of two to account for dynamic amplification.

$$Force = 2 \times \rho_l \times V_{eff}^2 \times A = 2376 \text{ lbf} \qquad (1)$$

The effective velocity (16.8 ft/sec) was calculated as the sum of the superficial gas velocity (14.4 ft/sec) and the superficial liquid velocity (2.4 ft/sec).

FIG. 3. Detached support flange near the top of the sphere.

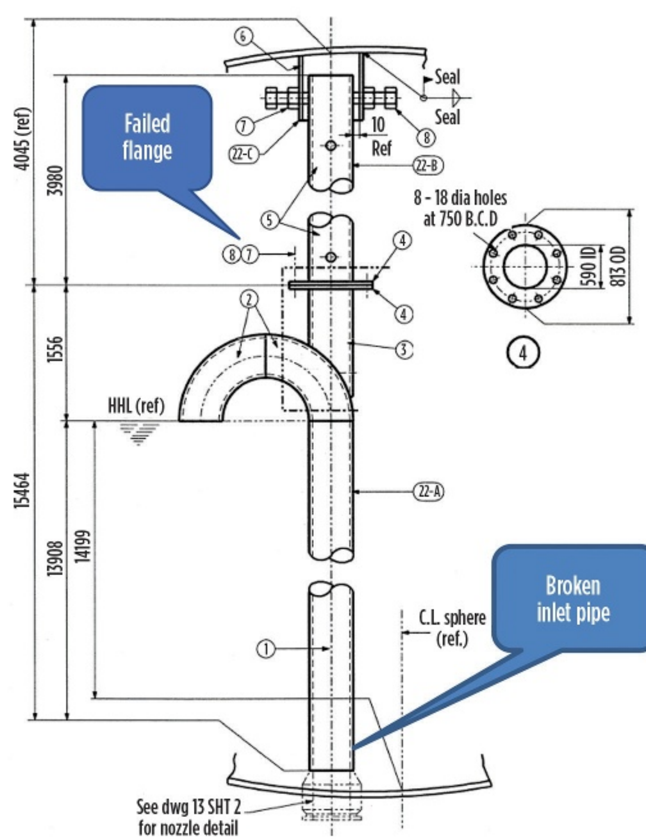

FIG. 4. Inside the sphere, the inlet pipe was detached.

Inlet pipe fatigue analysis from slug flow. A proprietary stress analysis program[a] was used to calculate inlet pipe fatigue stresses. The missing guide support would have caused a fatigue failure at the inlet nozzle. No failure is expected with an active guide support. The calculated stress is multiplied by two to account for peak stress, as per ASME NC-3673.2. The resultant peak stress of 17,139 psi is placed into the ASME VIII 3.F.2 fatigue curve for un-welded carbon steel. With approximately 200,000 cycles, if slugs can occur every minute or so, the inlet pipe failure can happen in approximately 4.6 mos. The stress at the pipe support weld to the top of the vessel was much less, at 1,040 psi, so no failure is expected at that location (**FIG. 7**).

Inlet pipe flange fatigue analysis. The stress analysis program was used to calculate the moment at the pipe support flange (**FIG. 8**) as 20,924 ft/lbf. Using the calculated slug moment, the bolt stress is calculated as per the ASME B31.8 equation. The calculations show that the bolt stress is satisfactory based on static loads (**FIG. 9**). For cyclic loads, the bolt stress range is 27.6 ksi with a fatigue life of 70,000 cycles.

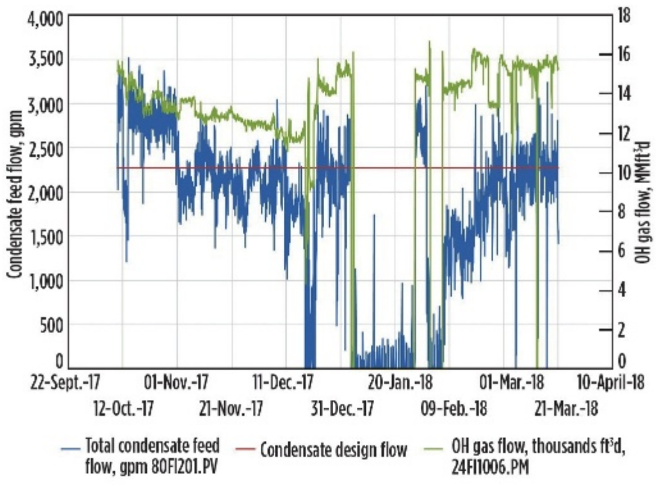

FIG. 5. Process data findings.

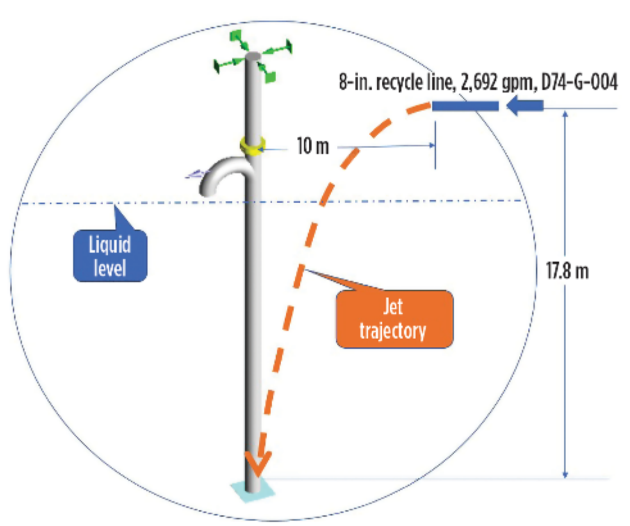

FIG. 6. Jet flow trajectory calculations show that the jet has little effect on the inlet pipe.

The 24-in. support flange bolting is shown to scale in **FIG. 10.** Problems with this flange design included:

- Bolts too far from pipe, causing prying and bending of plate
- Thin plate (12 mm)
- Small bolts (⅝ in.)
- Few bolts (8)
- Single nut is used.

To strengthen the old design, a new support flange design, shown in **FIG. 11,** was proposed. Advantages include:

- Bolts are closer to the pipe, eliminating prying effect
- Thicker A36 plate (18 mm vs. 12 mm) and smaller radius (760 mm vs. 813 mm)
- Larger bolts (1 in.)
- More bolts (16)
- Double nut with tack weld to avoid loosening from vibration.

Based on the calculation,[1] the bolt stress is reduced from 27.6 ksi to 5.96 ksi (a reduction of 4.6 times). The fatigue life is increased from 70,000 cycles to 10 MM cycles, or 142 times (**FIG. 12**).

Assumptions. Slug loads are based on gas and liquid rates that are measured at the outlet pipes. Inlet flow may be higher, but average values are the same. A dynamic amplification factor of 2 is used in calculating the slug forces. The slug force is

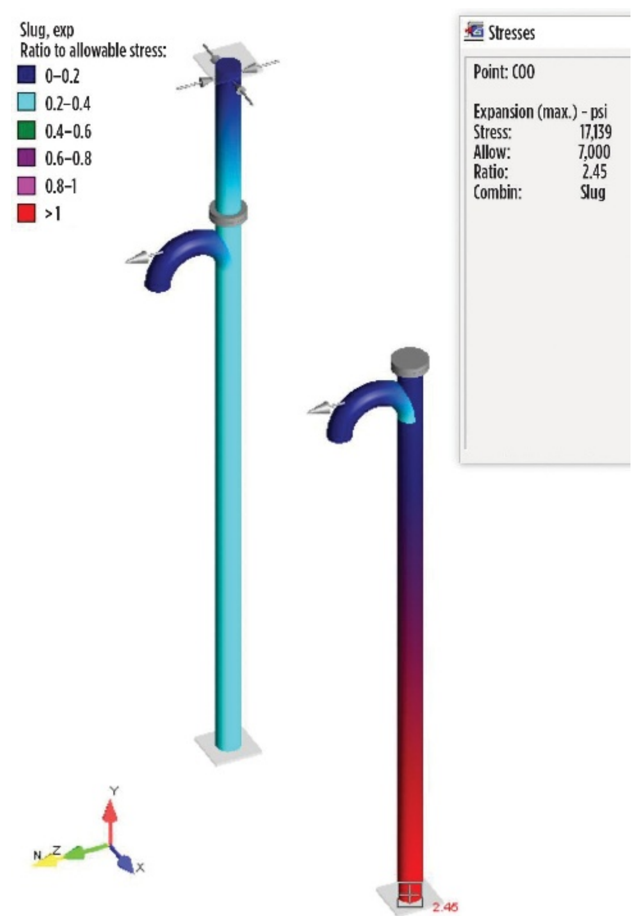

FIG. 7. Since the stress at the pipe support weld to the top of the vessel was much less at 1,040 psi, no failure is expected at that location.

based on the highest possible force, assuming an all-liquid slug is pushed by the gas.

Analysis summary:

- Jet flow from the 8-in. recycling line would not impact the inlet pipe and, therefore, cannot be a cause for the failure.
- The weak flange connection with eight small bolts failed by overstressing due to high cyclic slug loads caused by continuous use of the sphere as a stabilizer. Bolt loosening could have triggered an earlier failure, as well.
- High condensate flowrate to the sphere beyond the original design rate of 2,275 gpm (78,000 bpd) was observed on several occasions and was more frequent during October 2017–March 2018 (FIG. 5). The flow reached 3,500 gpm (120,000 bpd), particularly in October 2017, when the average flow was 3,200 gpm (110,000 bpd).
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- When one stabilizer is out for cleaning, the sphere is used as a backup stabilizer. During this scenario, the piping configuration preferentially diverts more flow to the sphere, causing the flow to exceed the sphere design capacity.

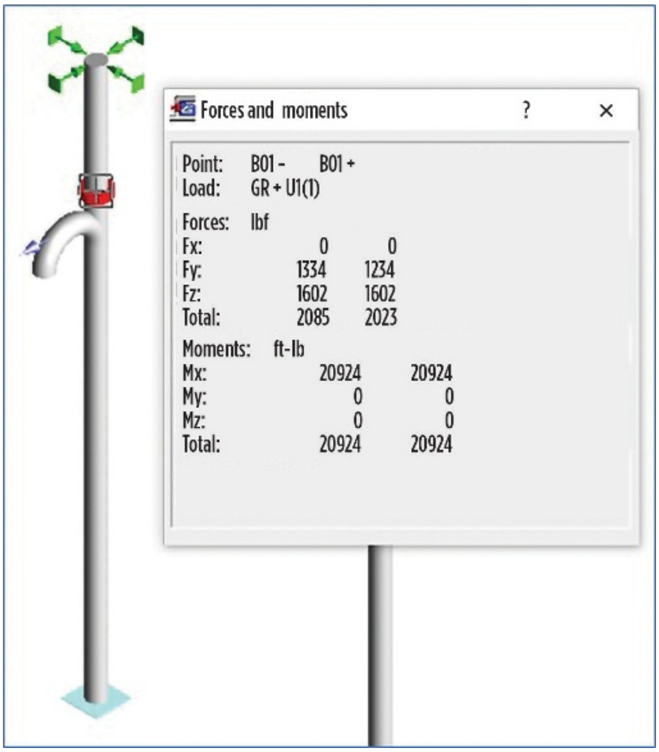

FIG. 8. Flange moment from slug forces.

Check existing bolts

per ASME B32.8 equation	$M_L = (C/4)(S_b A_b - PA_p)$
Calculated moment at flange	$M_L = 20900 \times ft \times lvf$
Bolt size	$d_b = \frac{5}{8} \times in.$
Number of bolts	$N = 8$
Total bolt area	$A_b = N \times \pi \times \frac{d_b^2}{4}$
Bolt circle diameter	$C = 750 \times mm$
Bolt stress	$S_b = \frac{4 \times M_L}{C \times A_b} = 13.843$ ksi
Allowable stress (SA-193 B7M) B32.3	$Sa = 25 \times ksi \quad S_b < Sa$

Static bolt stress check passed

Fatique check:	
Bolt stress range	$2 \times S_b = 27.685$ ksi
Estimated cycles (UT-Austin tests)	Cycles = 70,000

Check new bolts

Bolt size	$d_b = 1 \times in.$
Number of bolts	$N = 16$
Total bolt area	$A_b = N \times \pi \times \frac{d_b^2}{4}$
Bolt circle diameter	$C = 680 \times mm$
Bolt stress	$S_b = \frac{4 \times M_L}{C \times A_b} = 12.982$ ksi
Bolt stress range	$2 \times S_b = 5.964$ ksi
Estimated cycles (UT-Austin tests)	Cycles = 10×10^6

FIG. 9. Based on static loads, the bolt stress is satisfactory.

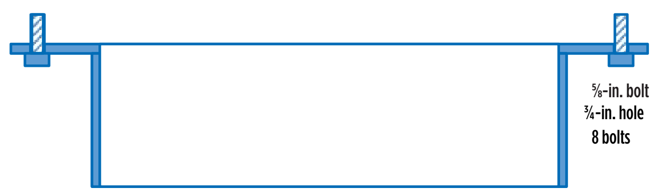

FIG. 10. Existing flange with bolting to scale.

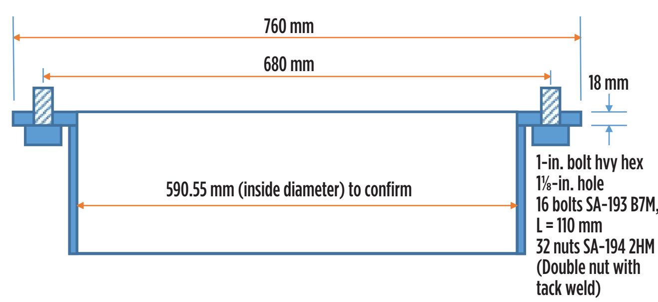

FIG. 11. Proposed support flange design.

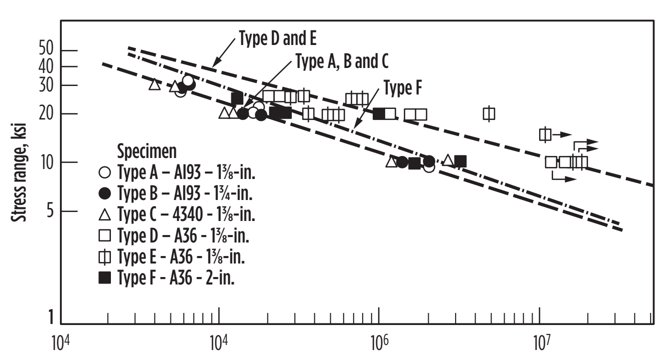

FIG. 12. Fatigue life calculations for A193 bolts.

Recommendations include:

- Replace the support flange to increase the fatigue life of the inlet pipe. The bolt stress will be reduced by 4.6 times, and a double nut with tack weld will prevent bolt looseness. Both flanges should be identical on both sides of the support. Plate material is A36, as per original design. Alternate plate material A516 grade 60 is acceptable. Plate thickness of 20 mm is also acceptable.
- Maintain the flow within the design rate of 92,000 bpd for liquid and 17 $MMft^3d$ for gas for two-phase flow until it can be confirmed that higher flows can be processed through a more detailed analysis with designer involvement.
- Temporarily avoid scraping operation on the pipelines when the sphere is utilized as stabilizer during the shutdown of one condensate stabilizer. This is to avoid sudden liquids surge to the sphere and results in high flowrate.
- When the sphere is utilized as a stabilizer during the shutdown of one condensate stabilizer, divert the condensate from the slug catchers directly to the sphere to minimize the impact of the imbalance in the gas/condensate ratio between the sphere and condensate stabilizers. During this mode of operation, ensure all slug catchers' water interface level controls are functioning and in auto mode to avoid water carryover.
- Inspect the strainers of the condensate and water pumps to locate the missing bolts for analysis. HP

NOTES

[a] Bentley's AutoPIPE stress analysis program

ACKNOWLEDGMENT

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LIERATURE CITED

[1] Fischer, F. L. and K. H. Frank, "Axial tension fatigue strength of anchor bolts," Center for Highway Research, The University of Texas at Austin, Research Report Number 172-1, March 1977.

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